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Dynamic Behavior of a Thin Disc in a Thrust Bearing Clearance: Effects of Solid and Fluid Friction

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ABSTRACT

A thin disc (washer) in a thrust bearing clearance is conceived as a improvement in the energy efficiency of the thrust bearing. This concept has been applied to reciprocating compressors. However the effect of frictional behavior is not fully understood and the reliability aspects are in question. This paper addresses the dynamics of the washer in the clearance considering both solid and fluid friction.

NOMENCLATURE

c	washer journal clearance
c_1	clearance between the washer and upper bearing plate
c_2	clearance between the washer and lower bearing plate
e	eccentric of the shaft
F	washer journal load
h	washer height
M_f	frictional moment due to the washer face friction with thrust faces
m	mass of the washer
M_b	moment on washer due to the washer journal oil film
N	revolutions per minute (rpm)
p	pressure on the thrust washer
R_i	washer inside radius
R_o	washer outside radius
R	thrust bearing shaft radius
T	total time interval for one revolution
t	time
W	total weight (load) on washer
X,Y	coordinate system
z	viscosity-pressure index
ε	washer journal eccentricity
η	coefficient of friction
μ	viscosity of lubricant oil
μ_{new}	viscosity with pressure variation
ω	shaft angular velocity
$\omega_w = \dot{\psi}$	washer angular velocity
ϕ	attitude angle
$\ddot{\psi}$	acceleration of the washer
ρ	washer density
θ	shaft angular displacement
Δt	time interval

1. INTRODUCTION

Thrust bearings are one of the most commonly used bearing configurations in the industrial applications utilizing parallel disk configuration. The thrust bearing supports the shaft weight and other loads associated with the shaft. The shaft rotates and load of the shaft is supported due to lubricant pressure generation, although the pressure development mechanism is not fully understood. A different operating condition can be arrived at by introducing a thin washer in the thrust bearing (Figure 1). As the shaft rotates the top face of the thrust bearing rotates while the bottom thrust face is static. With the introduction of the washer, the washer top face is subjected to the moments due to the shaft rotation while the bottom face is devoid of any forcing torque. However the restoring torques act on the washer due to the frictional moments due to the viscous friction or metallic friction.

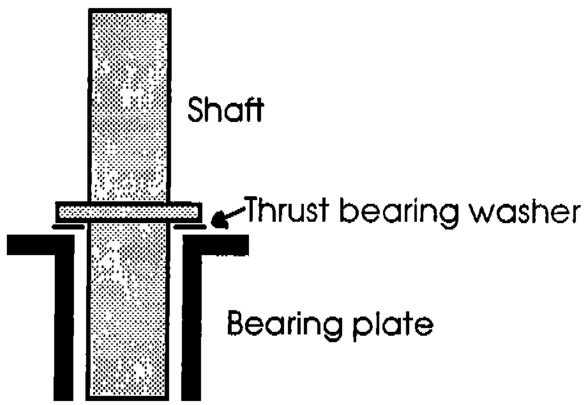

Figure 1. Schematic diagram of the thrust bearing washer configuration

To this authors knowledge there is no prior publication in this interesting phenomena. The design has been used extensively in past by a compressor manufacturer to increase energy efficiency of the compressor.

This paper presents the mathematical model and solution procedure for the dynamics of the washer. The variation of friction coefficient at washer and thrust faces are also analyzed. The understanding of the washer dynamics can help in designing a more reliable thrust bearing as well as in reducing the bearing and other mechanical losses to increase the compressor efficiency.

2. PRESENT WORK

In the present work, the dynamics of the washer is described in detail. The dynamics is solved using the numerical methods. This is carried out for several rotations of the shaft until steady state velocity is reached by the washer. For the analysis, a Cartesian coordinate frame is attached to the center of the shaft.

2.1 Washer Dynamics

The dynamics of the washer is governed by the torques exerted on it. The various forces and torques that influence the washer motion include the washer journal bearing moment, and frictional moment at the washer ends. A relative velocity between the washer and shaft can be calculated and the washer can be modeled as stationary for the washer journal moment calculation. Taking moment about the washer center the equation of motion for the washer is as follows (Figure 2):

$$I\ddot{\psi} = M_b + M_f$$

where M_b is the journal oil film moment on the washer, M_f is the moment due to friction between the

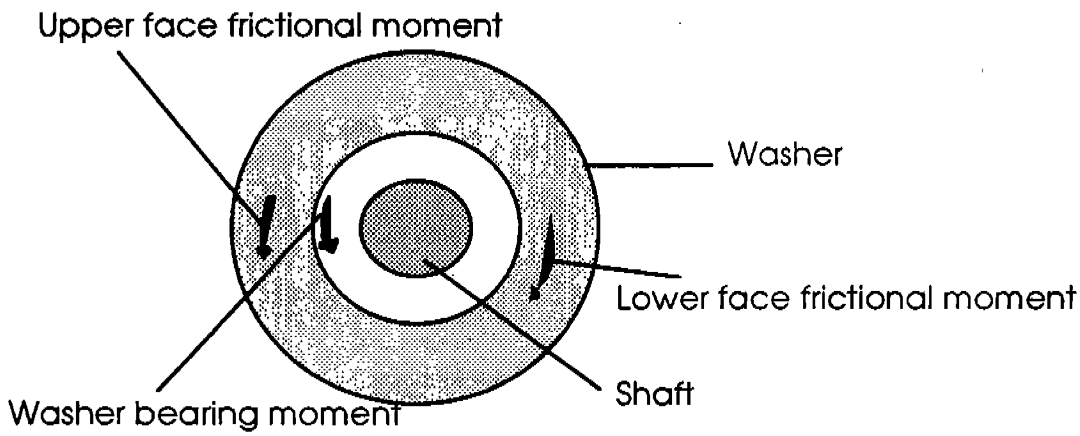

Figure 2. Washer dynamics

washer ends and the bearing plates. I is the mass moment of inertia about the polar axis and is given by

$$I = \frac{1}{2}m\left(R_o^2 + R_i^2\right) = \frac{1}{2}\pi\rho h\left(R_o^2 - R_i^4\right)$$

2.1.1 Washer journal bearing moment

The washer journal bearing is treated as an short journal bearing. Considering the washer as a stationary bearing, the washer journal moment is given by

$$M_b = M_r - F\varepsilon c \sin\Phi$$

where ε is the bearing eccentricity ratio and is given by $\varepsilon = e/c$, Φ is the attitude angle (i.e., the angle through which the load vector has to be rotated in the direction of the journal rotation to bring it into line of centers and F is the washer journal load). The oil film moment due to the rotation of the shaft M_r, is given by [1]

$$M_r = \frac{\mu\pi R^3 h\left(4+5\varepsilon^2\right)(\omega-\omega_w)}{c\left(2+\varepsilon^2\right)\sqrt{1-\varepsilon^2}}$$

The journal load on the thrust bearing washer is zero and hence the bearing eccentricity in the washer journal bearing is zero. Then the oil film moment on the washer becomes

$$M_b = \frac{2\mu\pi R^3 h(\omega-\omega_w)}{c}$$

2.1.2 Friction moment at washer ends

Assuming viscous friction, the friction moment on upper face of the washer is derived as follows:

$$M_{f1} = \mu\left[\int_{R_i}^{R_o} 2\pi r dr \frac{r(\omega-\omega_w)}{c_1} r\right] = \frac{\pi\mu(\omega-\omega_w)\left(R_o^4 - R_i^4\right)}{2c_1}$$

and for the lower face the moment becomes

$$M_{f2} = \mu\left[\int_{R_i}^{R_o} 2\pi r dr \frac{r\omega_w}{c_2} r\right] = \frac{\pi\mu\omega_w\left(R_o^4 - R_i^4\right)}{2c_2}$$

When the lower end of the washer is in contact with the lower bearing plate, then metallic friction can be assumed. In this case, the total moment will consist of a term with metallic friction and another term with viscous friction. The metallic friction term is given by

$$M_{f2}\Big|_{met} = \frac{2}{3}\eta\ W\frac{\left(R_o^3 - R_i^3\right)}{\left(R_o^2 - R_i^2\right)}$$

The total friction moment on the washer becomes

$$M_f = M_{f1} - M_{f2}$$

Incorporating the sign for the friction forces, the equation can be written as

$$M_f = E_1(\omega - \omega_w) - E_2\omega_w - E_3$$

where E_3 and E_4 are given by

$$E_1 = \begin{cases} \frac{1}{2c_1}\pi\mu\left(R_o^4 - R_i^4\right) + \frac{2\mu\pi R^3 h}{c}, & \text{for viscous friction on both faces} \\ \frac{1}{2(c_1+c_2)}\pi\mu\left(R_o^4 - R_i^4\right) + \frac{2\mu\pi R^3 h}{c}, & \text{for upper face viscous friction} \end{cases}$$

$$E_2 = \begin{cases} \frac{1}{2c_2}\pi\mu\left(R_o^4 - R_i^4\right), & \text{for viscous friction on both faces} \\ 0, & \text{for upper face viscous friction} \end{cases}$$

$$E_3 = \begin{cases} 0, & \text{for viscous friction on both faces} \\ \frac{2}{3}\eta\ W\ \frac{\left(R_o^3 - R_i^3\right)}{\left(R_o^2 - R_i^2\right)}, & \text{for upper face viscous friction} \end{cases}$$

In addition to the rotational component of friction, there is a sliding component of friction. However, this part of the friction force goes through the center of the washer and the moment due to this part of friction force is zero.

Then the equation of motion for the washer becomes

$$I\ddot{\psi} = E_1(\omega - \omega_w) - E_2\omega_w - E_3$$

2.2 Effect of load on viscosity variation

The thrust bearing loading consists of the weight of the shaft and the electric rotor. The viscosity of the lubricant oil varies with the pressure and temperature, and is different at the thrust bearing than the oil sump. Roeland's formula is used to calculate the operating viscosity at the thrust bearing clearance. For isothermal conditions the equation is given by [2]

$$\log_{10}\mu_{new} = \left(\log_{10}\mu + 1.2\right)\left(1 + \frac{p}{2000}\right)^z$$

where viscosity is in centipoise, pressure is in kgf/cm^2 and the viscosity-pressure index z is a dimensionless constant and has a value of 0.48 for advanced ester and 0.67 for super refined naphthenic mineral oil.

2.3 Solution procedure

The algorithm for the solution procedure is given in Table 1 [3]. The Runge-Kutta method of solutions yields values for ω_w. The two ordinary differential equations that govern the washer dynamics are presented here again for convenience.

$$I\ddot{\psi} = E_1(\omega - \omega_w) - E_2\omega_w - E_3$$

$$\dot{\theta} = \omega$$

Initial conditions:

$$\omega|_{t=0} = \frac{2\pi N}{60},\ \left(\dot{\psi} = \omega_w\right)\big|_{t=0} = 0,\ \psi|_{t=0} = 0,\ \theta|_{t=0} = 0$$

These equations are solved by fourth order Runge-Kutta method. Small time steps are used to carry out the iteration. The solution starts with an assumed zero velocity for the washer. To yield a steady state solutions a convergence criteria is imposed on the iteration procedure. This criteria is given as follows:

$$\left|\omega_w|_{new} - \omega_w|_{old}\right| \le 1\times10^{-5}$$

Table 1. Solution Algorithm

```
For convergence of solution iterate the whole process M times
    Do I = 1,M
        Time span for one cycle: T = 60 / N ,
        Divide time span to J intervals
        Initialize
        ω = 2πN/60, θ = 0,
                Do K=1,J
                    Washer Dynamics
                        Calculate E1, E2, E3, I
                        Solve for ψ̈ , ψ using Runge-Kutta method
                        Calculate velocity of washer
                    Advance the time step
                End do
    End do
```

3. DISCUSSIONS

The washer in the present analysis is a thin washer and the washer journal clearance is significantly higher that the thickness of washer. For practical purposes, the washer journal effects are negligible. The washer rotates in the same direction as that of the thrust bearing when there is viscous friction in both upper and lower faces. However the washer is stationary if the lower face friction is metallic. If due to asymmetry, the washer is in contact with the shaft, and the lower face friction condition of the washer is metallic then the washer can act as a knife and cut the shaft. This theory has been validated by the friction characteristic at the thrust bearing and washer interface. It is found that for compressors using R-134a refrigerant and ester oil as lubricant, the friction coefficient is high and unstable. The scattering friction force behavior can also induce irregular motion to the thrust bearing washer. In case of R-12 and mineral oil combination the friction coefficient is stable. In addition the lack of chlorine molecules in the R-134a system also deprives the thrust bearing of boundary lubrication. This can lead to a metallic contact between the thrust bearing and washer. Quantifiable number are not presented due to the proprietary nature of the data.

4. CONCLUSION

This paper presents a complete theoretical treatment of the washer dynamics in a thrust bearing. Employing Runge-Kutta method the washer angular velocity is calculated. It is found that the washer remains stationary if the lower face of the washer is in metallic contact with the lower thrust bearing surface. The washer rotates in the same direction as that of the shaft when the washer is in viscous friction. The magnitude of the rotation varies with the clearance variation and the distribution of the clearance in upper and lower face of the washer. The magnitude variation is found to be linear with the variation of upper face clearance. With the increase in upper face clearance the rotational speed of the washer decreased. The washer can act as a knife when it is thin, and the lower face is in metallic contact with the thrust bearing stationary surface. If due to asymmetrical configuration, the washer is in contact with the shaft, the washer can cut the shaft and lead to thrust bearing failure. The summary of conclusions is described in table 2.

Table 2. Summary of conclusions

Refrigerant + Lubricant	Lubrication condition at lower thrust face and washer	Comments
R-12 + Mineral oil	Well lubricated and stable, good boundary lubrication	No shaft cutting
R-134a + Ester oil	Metallic and unstable, poor boundary lubrication	Shaft cutting

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